# Metrolink Glendale Collisions
# CPUC Data Collected

## Development of a Passenger Rail Vehicle Crush Zone

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# DEVELOPMENT OF A PASSENGER RAIL VEHICLE CRUSH ZONE

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**ABSTRACT**
The use of crush zones in passenger rail vehicles is rapidly growing in the United States and throughout the world. Such crush zones are an important part of the crash energy management philosophy of train occupant protection. The objective of this study was to determine the advantages, disadvantages and issues related to incorporating crush zones at the ends of coach cars for protection in collisions between two trains. The general specifications for the crush zone were selected after consideration of the energy and forces that can be accommodated in such structures. Various designs were considered to meet these requirements and one of these was selected for more detailed development and evaluation. The effort included design layout and nonlinear dynamic finite element analysis to determine crush response.

## INTRODUCTION

There is a general trend in the development and construction of passenger rail vehicles throughout the world today known as crash energy management design. The basis of this philosophy is, on the one hand, an explicit acknowledgement that each train collision has associated with it a collision energy that must be dissipated and, secondly, that the greatest safety to train occupants and others is through a deliberate, planned control or management of this energy. Although this philosophy has been used in automobiles for decades, it is only within the last few years that it has had widespread application for trains.

Crash energy management design philosophy generally includes the selection of a collision scenario or scenarios against which protection is to be provided, including the specific desired outcome. A common example is a collision between two like trains, one moving, the other stationary, in which the ends of passenger-containing vehicles are not to crush more than, say, 36 inches (0.9 m) with a peak vehicle deceleration of less than 6 g's. The collision scenario automatically defines the collision energies that must be managed as well as several other key parameters.

A key principal of the crash energy management philosophy is to absorb energy at locations in which there are no occupants. The locations could range from 'sacrificial' baggage cars to the more common approach of absorbing the collision energy in the normally unoccupied ends of several rail cars. This latter approach is sensible when one considers, (a) the importance of maximizing total passenger space for revenue-generating service, (b) the large energies to be absorbed, especially in train-to-train collisions, and (c) the need to preserve the normally occupied spaces during a collision.

While current orders for trains with crash energy management systems provide tangible evidence of their practicality, there is still a lack of public knowledge about some of the implementation details and weight and cost penalties associated with this new design philosophy. The work described here had as its overall objective a determination of the issues and possibilities of incorporating crush zones at the ends of passenger rail vehicles. The crush zone described here was developed for intercity coach cars but is also applicable for other types of service.

## CRUSH ZONE DEVELOPMENT

### Specifications

The specifications for coach car crush zones include energy absorption capability, maximum crush force, maximum crush distance, and various strength requirements for handling normal operation and for the prevention of override and lateral buckling. The specifications are generally derived from the collision scenario against which protection is to be provided and from current knowledge about the conditions of crush and deceleration below which occupants will avoid serious injury. In some cases, the specifications differ for different vehicles in a train consist, as determined by lumped-mass-type collision dynamics models. The basis for the specifications used here are described in a separate publication [1], although some explanation is provided here.

For example, we chose a coach car energy absorption goal of $1.5\times10^6$ ft-lbf (2MJ), based in part on a particular train-to-train collision scenario and on what practice has shown to be feasible. As a reference, new British Rail commuter coach cars are required to absorb $0.75\times10^6$ ft-lbf (1MJ) of energy at each end [2].

The maximum absorbable energy at a coach car end can be estimated for a particular set of conditions. The absorbed energy can be expressed as the product of the average crush force and the extent of crush:

$E_{abs} = (F_{avg})(l_c)$.

If we place constraints on the magnitude of the crush force and the extent of crush, we automatically limit the amount of energy that can be absorbed. The maximum crush force is determined by the car body strength, which is generally related to the buff strength of the vehicle, and by the need to limit the decelerations experienced by occupants during a collision.

For example, suppose we wish to limit with certainty the average acceleration to 6 g's and the amount of crush to 3 ft (0.9 m) in a coach car with an $800\times10^3$ lbf (3.6MN) buff strength. Then, for a 100,000 lbm (45,300 kg) car, the maximum allowable load is about $600\times10^3$ lbf (2,670 kN), which is consistent with the buff strength, and the energy absorption is:

$E_{abs} = (600{,}000)(3) = 1.8\times10^6$ ft-lbf (2.44 MJ).

Thus, our choice of a $1.5\times10^6$ ft-lbf (2 MJ) goal is below but close to the maximum energy absorption capability based on such considerations.

These and the remaining specifications used for the design of our coach car crush zone are listed in Table 1.

**Table 1. Coach Car Crush Zone Specifications Used in this Study**

| Parameter | Value |
|---|---|
| Energy absorbed | $1.5\times10^6$ ft-lbf (2 MJ) |
| Design crush length | 2-3 ft (0.6-1 m) |
| Push-back coupler force | $600\times10^3$ lbf (2,670 kN) |
| Mean crush force | $700\times10^3$ lbf (3,110 kN) |
| Peak crush force | $800\times10^3$ lbf (3,560 kN) |
| Vertical strength (during crush) | $200\times10^3$ lbf (890 kN) |
| Permissible lateral deviation of longitudinal crush load center of action from vehicle center line | ≥12 inches (0.3 m) |
| Permissible vertical deviation of longitudinal crush load center of action from underframe | +18 inches (0.5 m) |

A pushback coupler was specified to ensure direct interaction of underframes in a severe collision and to promote greater distribution of energy absorption among vehicles via a slack effect [1,3]. A pushback force of $600\times10^3$ lbf (2,670 kN) appears to be high enough to prevent the pushback mechanism from operating (and thus to prevent the need for repair) for all but the most severe impacts. A peak crush force of $800\times10^3$ lbf (3,560 kN) was used with the expectation that decelerations would be lower than the theoretical maximum (because loads are applied to both ends of the vehicle at approximately the same time) and that the average crush force would also be lower. Although not common in crash energy management specifications today, we also felt it was important to include a permissible lateral deviation of longitudinal crush load center-of-action from the vehicle center line. This is because actual train collisions will not occur against a rigid wall and it is likely that there will be some deviation of the line of action between coupled cars. The value of 12 inches (0.3 m) was selected without technical analysis as a large but reasonable potential deviation. We also used as a specification a permissible vertical deviation of longitudinal crush load center of action from the underframe. Although the crush zone will be designed to prevent override, we felt that the crush zone should nevertheless be capable of absorbing the design energy if loads were applied to the collision posts above the underframe.

**Existing/Planned Strategies**
There are now several rail vehicle systems planned or in operation throughout the world that include crash energy management systems. Some of these are listed in Table 2 together with some of the characteristics of the coach cars. As of this writing, some new purchases of other transit vehicles are also requiring crush zones at the ends of coach cars.

**Crush Zone Concept**
The development of a coach car crush zone concept included review of existing systems (see Table 2) and idea generation sessions. We selected concepts that could potentially be adapted to the type of underframe design that is currently found in practice in North America. Such construction consists of a steel underframe to which is attached a stainless steel, steel or aluminum superstructure consisting of light section frames and purlins and skin, all of which participate in load transfer. In particular, we used the Amfleet II intercity coach car as a base from which to make modifications. Three different crush zone concepts were laid out (see [1]) from the various ideas that were generated. From these, only one was selected for detailed consideration.

Figure 1 shows the conceptual drawing and Figure 2 shows the finite element mesh developed to simulate this design; the latter is a one-half model, turned on its side to expose the structure on the underside of the vehicle.

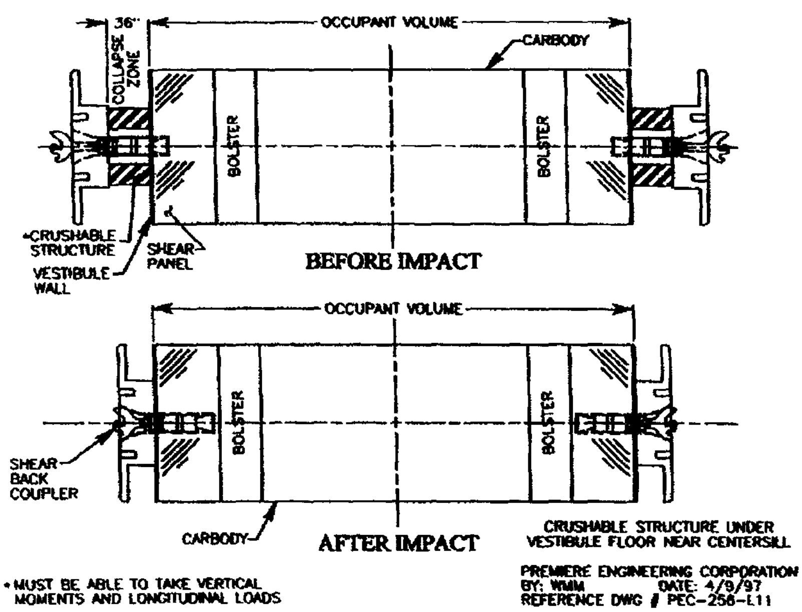


**Figure 1. The Concept Layout for the Sliding Sill, Double Crush Element Coach Car Crush Zone**

Table 2. Examples of Crush Zone Characteristics for Existing/Planned Coach Cars

| Vehicle | Organization | Status | Design Energy Absorption (kJ)* | Material | General Crush Element Characteristics | References |
|---|---|---|---|---|---|---|
| ORE Research Vehicle | British Rail | Built and tested | 1,000 | Steel | • Aluminum honeycomb<br>• Fiber composites | 1 |
| BR Mk1 (Research Vehicle) | British Rail | Built and tested | 1,500 | Steel | • Steel, large cell honeycomb<br>• Rectangular box sections | 2 |
| 465 Networker | Adtranz | Built and tested | 1,000 | Aluminum | • Aluminum honeycomb<br>• Rectangular box sections | 3 |
| TGV 2N Extreme Trailer | SNCF | In service | 3,500 | Aluminum | • Slotted box beams<br>• Thin-walled prismatic box sections | 4 |
| XTER<br>Front | SNCF | Built and tested | 5,000 | Stainless | • Crushable composite in draft gear<br>• Outboard prismatic box sections<br>• Inboard prismatic box sections | 6 |
| American Flyer Trailer Car<br>*Adjacent to Power Car*<br><br>*Not adjacent* | Bombardier | In design phase; energy absorbers tested | <br><br>5,000<br><br>2,000 | Steel | • System is likely to be similar to TGV systems<br>• Structure must also satisfy North American requirements, including S-580 (power car) and 49 CFR, Part 229.141 (both) | 7 |
| NJT LRV | Kinky-Sharyo | In design phase; energy absorbers tested | 400 | Steel, with aluminum absorbers | • 350kJ must be provided by vehicle structure | 8 |
| NYCT | Bombardier, Kawasaki | Just awarded | 1,000 | Steel | • Energy absorption to be provided by controlled deformation of the car body | 9 |

* At one end.

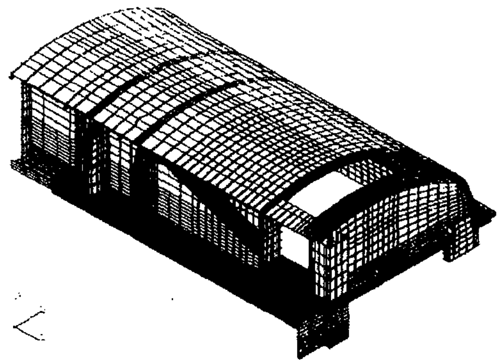

**Figure 2. The Finite Element Model Used for the Coach Car Crush Zone**

The energy absorption for this concept occurs primarily in two 'double-box' crush elements (see below), one on each side of the center sill. The center sill at the ends of the car is a sliding sill, in which one rectangular box section is permitted to slide into another similar fixed sill after a set of bolts has been fractured in shear from the high collision loads. Thus, the sliding and fixed sills provide a load path for normal operating loads and for bending loads during crush of the absorbers; the energy absorbers only carry load during the crush event. Bending loads are carried through contact between the sliding and fixed sills. The use of sliding sills is a common approach for energy absorption in many US freight cars.

The crush zone includes a pushback coupler that activates when another set of bolts is fractured by shear during the collision. This provides a degree of slack, as mentioned previously, as the coupler pushes back and it enables the anticlimbers and underframes of adjacent vehicles to interact directly during the collision event after the coupler has pushed completely back. Thus, this particular crush zone includes two 'triggers': the first to activate the pushback coupler and the second to activate the crush elements. Though not described here, this crush zone is designed to include ribbed anticlimber elements at each corner to assist in the prevention of override.

Other modifications were needed to obtain satisfactory crush performance for this crush zone. For example, a shear plate was added to the top of the underframe just behind the crush zone, and the sides and roof structure were longitudinally reinforced to prevent any significant local deformation of the occupant volume inboard of the vestibule wall. Reinforcements were also added on the underside of the underframe (see Figure 2) to support the back of the crush elements and the sides of the fixed sill. These modifications were made to an existing design layout similar to the Amfleet II coach car. Effort to redesign the entire end to optimize weight was beyond the scope of the project.

Thus, the weight added to the vehicle as a result of our modifications is relatively high: about 5,000 lbm (2270 kg) per vehicle. We believe that detailed optimization of these modifications alone would reduce this value by at least one-half. Furthermore, design 'from scratch' of a vehicle to include such a crush zone would likely reduce the weight increment even further. The existence of vehicles that contain crush zones whose weights are comparable to strength-based designs supports this assertion.

Finally, it is clear that the doors, if left within the crushable length of the vehicle end, as shown in Figure 2, would be severely deformed as a result of the crush. This would not be acceptable for escape purposes. Therefore, the doors would need to be located inboard of the crush zones, which could reduce potential passenger space for some types of operation.

**CRUSH RESPONSE**

We conducted our analysis of the response of the crush zone using the commercially available computer program ABAQUS/Explicit [4]. The models consist almost entirely of shell elements. The material model was represented by a piecewise linear stress-strain curve based on a material with yield and tensile strengths equal to 50 ksi (345 MPa) and 80 ksi (550 MPa), respectively. In general, simulations consisted of a flat, rigid mass moving at 30-60 mph (48-97 km/hr) colliding with the structure of interest.

**Crush Element Response**

The first step in the crush zone evaluation is the analytical verification of the crush element response. As mentioned, we selected a 'double-box' crush element, in this case made of steel, for the energy absorbing elements. Figure 3 shows its geometry. We kept the geometry of the double-box simple for modeling purposes. In practice, it is advantageous to include features,

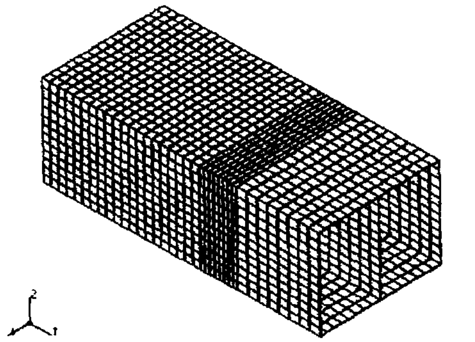

**Figure 3. Geometry of the Double Box Energy Absorber Used for the Coach Car Crush Zone: Width = 16 inches; Height = 8 inches; Thickness = 0.134 inches.**

such as reinforcements or localized deformations, to reduce the initial collapse load and to ensure that crush occurs in a similar manner for various collision speeds and load combinations. The rear edges of the crush element were fixed against all translations and rotations for the finite element analysis.

Figure 4 shows the energy absorption element load-crush response for a 60 mph (97 km/hr) collision with a 50,000 lbm (26,670 kg) mass. Figure 5 shows an example of the crush pattern from this analysis. There is an initial, high load peak, which corresponds approximately to the plastic buckling load for this section, but overall the crush load is quite uniform until the material begins to compact at about 20-25 inches (0.51-0.64 m) of crush. The energy absorbed in one element at a crush of 25 inches (0.64 m) is $0.77 \times 10^6$ ft-lbf (1.0 MJ.)

Fracture in this element is not predicted even though the maximum strain, which occurs in a fold at the junction between the center web and the outer surface, reached values close to 90%.

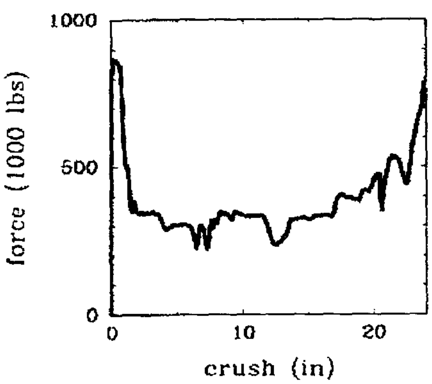


**Figure 4. Crush Response of the Double Box Energy Absorption Element**

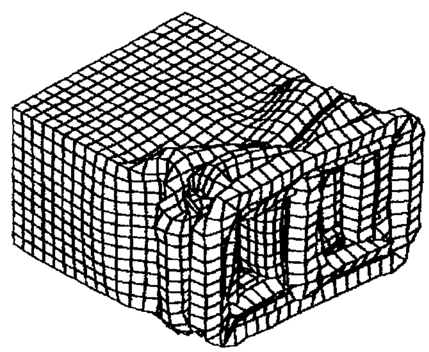

**Figure 5. Example of Double B Element Deformation: Crush Ω8 inches.**

**Crush Zone Response**

The analysis of the crush zone was carried out with the finite element mesh shown in Figure 2. The longitudinal center plane was thus treated as a plane of symmetry. A set of spring and mass elements were added to the rear of the model to represent the elastic stiffness and the mass of the rest of the vehicle, which had been determined separately from a model of the complete vehicle. Figures 6 and 7 show the deformed mesh and the load-crush response for the crush zone concept for a simulated 60 mph (97 km/hr) impact with a flat rigid surface having a mass of 100,000 lbm (53,340 kg.) The collapse is relatively uniform except for the roof structure, which we did not tailor for the crush zone. The peak crush load in this case, except for the short duration initial peak and the load after the crush zone has compacted, is approximately $700 \times 10^3$ lbf (3110 kN.) The energy absorbed at a crush of 26 inches (0.66 m) is approximately $1.5 \times 10^6$ ft-lbf (2 MJ) matching the initial requirements. The crush

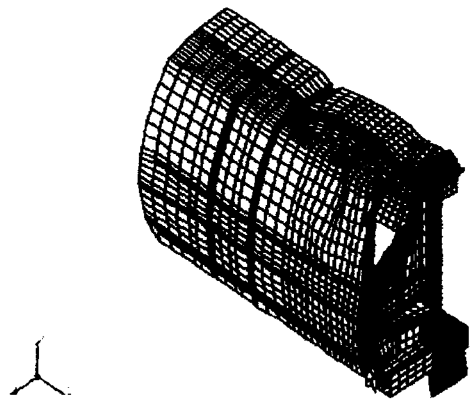

**Figure 6. Crush Zone Deformation after Approximately 24 inches of Crush**

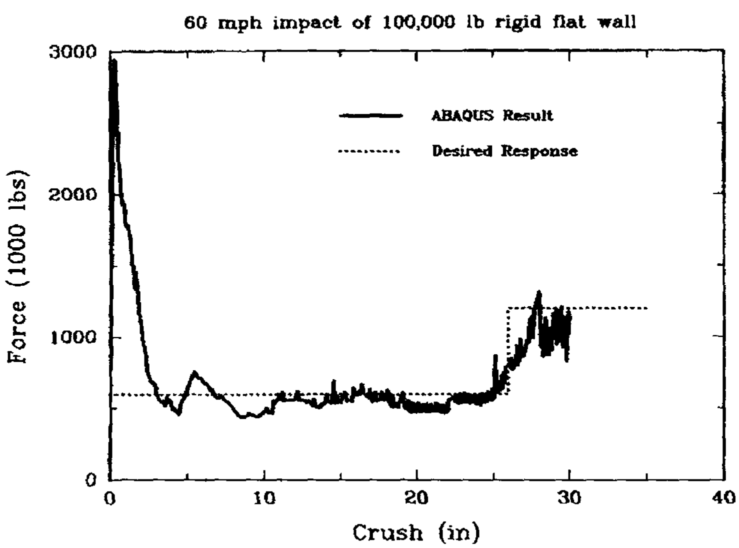


**Figure 7. Load-Crush Response of the Coach Car Crush Zone**

response for a simulated override loading was quite similar to that shown in Figure 7. However, the model collision post permitted intrusion into the occupied volume, due to excessive post bending. This suggests that stronger collision posts are required.

## SUMMARY AND CONCLUSIONS

This paper presents the conceptual development and the evaluation of a crush zone for intercity passenger rail vehicles for use in crash energy management systems in trains. It illustrates the many considerations needed in engineering such crush zones and offers a specific design that can be used for the type of vehicle underframe that is currently popular in the United States. Our analysis indicates that that a crush zone can be developed in an $800 \times 10^3$ lbf (3560 kN) buff strength rail car whose peak crush load is $700 \times 10^3$ lbf (3110 kN) and which absorbs $1.5 \times 10^6$ ft-lbf (2 MJ) of energy at each end. Furthermore, these performance specifications can be met for a variety of load locations and directions including override loads. The incremental weight associated with this crush zone design is significant. However, we believe that the weight penalty would be substantially lower if vehicle designers included the crush zone in the original design rather than modifying an existing design as we have done.

## ACKNOWLEDGMENTS

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## Evaluation of Cab Car Crashworthiness Design Modifications

James McInerney

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# EVALUATION OF CAB CAR CRASHWORTHINESS DESIGN MODIFICATIONS

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## ABSTRACT

A study was conducted to evaluate the effectiveness of structural modifications to rail cab cars for increased crashworthiness protection in train collisions. The crashworthiness benefits were calculated based on a particular design's ability to preserve the space occupied by the operators and the passengers during a collision. The influences of the modifications on vehicle weight and cost to manufacture were also estimated. The focus of the study was a collision scenario in which a cab car-led consist traversing a switch onto mainline track obliquely collides with a locomotive-led consist traveling in the opposing direction on the mainline track.

Modifying the strength of the end-structure members up to the load limits implied by the support structures – 800,000 pounds – increases the collision speed at which all the occupants are expected to survive to ~20 mph from ~10 mph for the baseline design. Within the allowable spaces of the baseline design, potential modifications have been developed which increase the end beam strength to nearly three times the baseline design strength, and increase the side sill strength to 1¼ times the baseline strength. Such design modifications, along with commensurate corner post and door post designs, made to the leading end of the cab car would add 670 lbs (~0.7% ) to the weight of the cab car and about $2000 (~0.1%) to the purchase price.

## INTRODUCTION

Recent train collisions in Secaucus, NJ and Silver Spring, MD have brought increased attention to the collision performance of cab cars. The potential for diesel multiple-unit (DMU) equipment, in which cab cars are used at both ends of the consist, to be used by a number of commuter rail authorities has also increased concern. In response, a study has been conducted to evaluate the effectiveness of structural modifications to rail cab cars for increased crashworthiness protection in train collisions. This paper describes some of the results from this study.

In a collision, any crushing of the front of the cab car results in a loss of occupant volume, with the potential for the operator and the passengers to be crushed. The principal objectives in providing crashworthiness are to preserve sufficient occupant volume in which the operator and passengers can ride out the collision, and to limit the forces and decelerations experienced by the occupants to survivable levels.

The focus of the study was a collision scenario in which a cab car-led consist traversing a switch onto mainline track collides obliquely with a locomotive-led consist traveling in the opposite direction on the mainline track. In such a collision, a relatively weak portion of the cab car strikes a relatively strong portion of the locomotive, leaving the cab car particularly vulnerable to structural crushing.

To establish a performance baseline, current cab car structural drawings were used to estimate the strength, weight, and force/crush characteristics of key components. Several modifications to the baseline end structure design were proposed to increase the strength of the cab car. (This study did not consider a redesign of the entire cab car structure.) Relatively simple modifications were considered to increase the area moment of inertia of key components, such as the collision posts, corner posts, end beams, door posts and side sills. The strength, weight, force/crush characteristics and cost for the modified designs were then estimated.

The design of U.S. passenger rail vehicles, with respect to crashworthiness and weight, has remained essentially unchanged for about 40 years. With the use of computational tools like finite-element analysis (FEA) which account for non-linear material behavior and allow large deformations, there is the potential to improve the rail equipment design in terms of crashworthiness, without adding weight to the vehicle, as has been accomplished by the automotive industry.

## SCENARIO DESCRIPTION

In February of 1996, two unrelated passenger train collisions occurred. In each collision, a cab car and a locomotive were the impacting cars. The initial impact occurred while one of the lead cars was traversing a switch, resulting in an angle between the cars, as well as a lateral misalignment. In both collisions, there was substantial crushing of occupant volume in the cab cars. The main load-bearing structural members of the cab cars (collision posts, draft sill, center sill) were not directly loaded during the collisions. Rather, weaker end members (end beam, side sill) were loaded directly, and failed. Figure 1 schematically depicts the initial conditions of an oblique collision.

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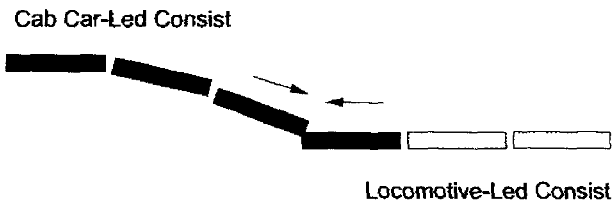



**Figure 1: Schematic of an Oblique, Cab Car-to-Locomotive Collision.**

The equipment involved in the Silver Spring collision had extensive damage, some of which appeared to be caused by override. The equipment involved in the Secaucus collision shows no indication of override. The collision scenario employed in this study is based on the collision that occurred at Secaucus, rather than Silver Spring, since a less detailed model – one that does not include the vertical motions of the cars – is required. Efforts to analytically evaluate a scenario that includes override, based on what occurred at Silver Spring, are ongoing.

The model was used to calculate the longitudinal, lateral, and yaw motions of the cab car and locomotive in an oblique collision using the calculated force/crush behavior of the end structure of the cab car. A description of the evolution of the Secaucus, NJ accident, developed from observations of the collision damage and discussions with accident investigators, is presented in Appendix A.

## STRUCTURAL DESIGN MODIFICATIONS

Cross-sectional properties of selected cab car end-structure components were modified in order to improve the structural crashworthiness of the vehicle. The structural members to be modified were chosen based on the consequences of the Secaucus collision. In order to limit the scope of the study, material choices were not varied from the existing design. Modifications to the main structure were not considered.

The typical North American passenger cab car structure consists of an underframe, an end structure and a body shell. The underframe consists of four longitudinal members: the center sill, draft sill and two side sills, and two body bolsters, which laterally connect the center sill, draft sill, and two side sills. The end structure consists of an end beam (or buffer wing), two vertical collision posts and two vertical corner posts. The structural members with the largest cross-sectional areas are the center sill, the draft sill, and the body bolster. The main cab car structural members are depicted schematically in Figure 2.

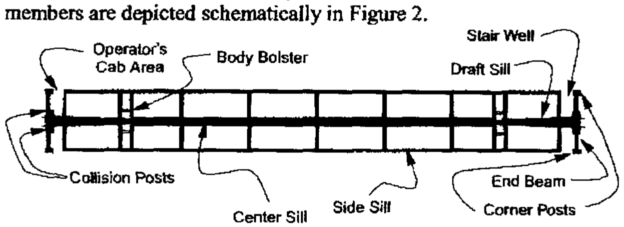



**Figure 2: Schematic of Typical Cab Car Structural Members, Top View.**

During a collision, the vertical members act to transfer loads to the underframe, particularly in the event of vertical misalignment or override between colliding cars. The addition of a second vertical member immediately aft of the stair well – a door post – may provide additional protection to the cab car passengers in an oblique collision in the event of vertical misalignment with a colliding vehicle. The body shell acts to stabilize the end structure and underframe members during a collision. Figure 3 illustrates the structural modifications considered.

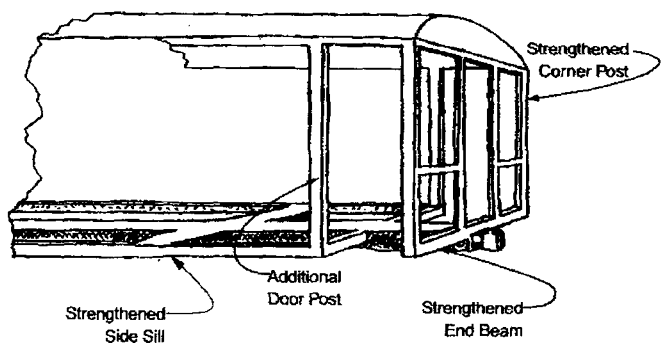



**Figure 3: Sketch of Cab Car Structural Modifications.**

Increased strength over existing member design was achieved principally by increased cross-sectional area. Modifications were generally made to the various components by replacing them with similar sections or rectangular tubes whose section moduli and/or cross-sectional area were greater than that of the existing component. In general, closed sections whose outer dimensions required little additional space over the existing component designs were selected. In some cases, higher component strength was accomplished by increasing only wall thickness. Table 1 lists the modifications considered. Two potential modifications were considered for both the door post and the side sill to better estimate the relationship between increases in strength, cost and weight.

**Table 1: Cab Car End Component Modifications.**

| Component | General Modification Description |
|---|---|
| Corner post | Aluminum rectangular tube section 10x6x0.5 inch with extra steel reinforcement at the base and a stronger end beam |
| End beam | Steel rectangular tube section with significantly greater wall thickness and welded reinforcement to draft sill |
| Door post | (1) Aluminum rectangular tube section 8x2.75x0.375 inch with stronger side sill and extra reinforcement at the base<br>(2) Aluminum rectangular tube section, 9x3x0.5 inch with stronger side sill and extra reinforcement at the base |
| Side sill | (1) Aluminum extrusion with significantly larger cross-sectional area along entire length to the vehicle<br>(2) Aluminum extrusion reinforced with another aluminum piece only to bolster |

The principal geometric constraint in modifying the cab car structure to improve crashworthiness is the location of the stair well immediately behind the end beam, which limits the locations for structural members. In addition, the floor plan of a typical cab car is almost entirely taken up by passenger and operator seating, leaving little unoccupied area to be designated as a crush zone. A typical cab car floor plan is shown in Figure 4.

**Figure 4: Typical Cab Car Floor Layout.**

Additional considerations in modifying the cab car structure to improve crashworthiness are weight and manufacturing cost. Increased manufacturing cost due to structural modification includes the cost of any additional material and/or additional labor to setup and weld. Increased weight can adversely affect the dynamic behavior of the cab car (the trackworthiness). Longitudinal and lateral balance of the car may influence the dynamic performance of the car. Increased weight may potentially lead to increased track and equipment maintenance. For example, track geometry may require greater maintenance to stay within tolerances and rails may wear more rapidly. On the equipment, brakes and suspension components may require more maintenance to remain within service limits. (Factors other than car weight may also influence track and equipment maintenance.) Taken to an extreme, increased carbody weight in turn may require larger brakes (and motors for MU equipment), which again increase total car weight. In discussions with various members of the industry, there appeared to be consistent opinion that increased car weight of less than 0.5% (about 500 lbs) would not adversely affect car trackworthiness nor measurably increase track and equipment maintenance as long as car balance was maintained, and that increased car weight up to 1% may be tenable without requiring other changes to the car, aside from those required to maintain balance.

The underframe, end structures, and body shell of a typical cab car account for approximately 20% of the car weight. (The interior furnishings – including the seats and wall panels – account for approximately 45% of the car weight, the trucks and suspension account for approximately 25%, other equipment including the draft gear, brake equipment, AC units, etc. accounts for the remaining 10%). As the entire car structure accounts for only a relatively modest portion of the car weight, significant increases in strength of end structure components may lead to modest increases in total car weight.

## STRENGTH CHARACTERISTICS

Four end structure components were selected for modification in this analysis. The primary measure of strength of each component in question was the load required to initiate yielding. The yield load for each component was calculated twice: once with the load applied at the floor level, and again with the load applied 18 inches above the floor level. The alternate load applications were used to evaluate the sensitivity of the components to bending loads. The equivalent axial load capacity was then determined as the product of the cross-sectional area and the yield strength. Crush energy at 1 foot of crush was also calculated for each component in order to compare the energy absorption capacity of the modified components with the baseline components. The crush energy is the integral of the force vs. crush curve.

The yield strength and crush energy for the baseline components are given in Table 2. Some of the strains calculated in the crush analyses were close to the fracture strains, suggesting that crush to 1 ft may not be achievable without fracture.

**Table 2: Strengths and Crush Energies for the Baseline Components.**

| | Strength @ Yield (kips)* | | Crush Energy @1 ft ($10^6$ in-lb.)* | |
|---|---|---|---|---|
| **Component** | **Load @ Base** | **Load @18 Inches** | **Load @ Base** | **Load @18 Inches** |
| Corner post | 110 | 45 | 1.6 | 1.2** |
| Door post | 75 | 60 | 5.2** | 2.6** |
| End beam | 110 | NA | 1.3 | NA |
| Side sill | 240 | NA | 5.2 | NA |

*For one such component.
**Fracture appears possible before reaching 1 ft crush.

Table 3 lists the yield strengths and crush energies for the modified components. The last column in the table indicates any weight added to supporting components in order to achieve the increased strength. Modifications to the end beam and side sill were required in order to increase the strength of the corner post and door post, respectively.

**Table 3: Strengths and Crush Energies for the Modified Components.**

| Strength @ Yield (kips) | | | | Crush Energy @ 1 ft ($10^6$ in-lbf) | | |
|---|---|---|---|---|---|---|
| Component | Total Weight (lbm) | Load @ Base | Load @ 18 Inches | Load @ Base | Load @ 18 Inches | Additional Components Weight (lbm) |
| Corner post | 140 | 300 | 115 | 3.6 | 2.8* | 150 |
| Door post(1) | 75 | 300 | 105 | 6.0* | 3.0* | 15** |
| Door post(2) | 100 | 300 | 140 | 6.2* | 3.9* | 15** |
| End beam | 260 | 300 | NA | 3.6 | NA | None |
| Side sill (1) | 820 | 300 | NA | 6.0+ | NA | None |
| Side sill (2) | 90 | 300 | NA | 6.0+ | NA | None |

*Fracture appears possible before reaching 1 ft crush.
**Based upon side sill modification 2.
+Based upon door post modification 1.

## WEIGHT AND COST ESTIMATIONS

The weight increase associated with a component modification was determined by the volume of material added to the modified component and any modified supporting elements (if necessary to carry the greater load). Cost estimates were based on the additional material, welding, fasteners, and set-up time for the component in question and any modified supporting components. The cost was estimated to the nearest $100. Table 4 lists the weight and cost increase for the proposed modifications, plus the ratio of the increase in strength and crush energy.

These results suggest that substantially strengthening the corner post results in a weight increase of 500 lb per vehicle for a 2.6-2.7 increase in strength ratio for the corner posts, over the baseline. Strengthening the corner posts and end beam, along with strengthening the door post and side sill on both sides of the front end of the car would add approximately 670 lbs to the weight of the car. The total cost of such modifications would be approximately $2000.

**Table 4: Summary of Weight and Cost Increases Associated with Strength and Crush Energy Increases.**

| | | | Strength Increase Ratio (lb.)* | | Energy Absorption Increase Ratio @ 1 ft Crush | |
|---|---|---|---|---|---|---|
| Component | Weight (Increase) (lb.)* | Cost** Increase ($US) | Load @ Base | Load @ 18 inches | Load @ Base | Load @ 18 inches |
| Corner post | 250 | 300 | 2.7 | 2.6 | 2.3 | 2.3 |
| Door post (1) | 40 | 200 | 4.0 | 1.8 | 1.2 | 1.2 |
| Door post (2) | 65 | 300 | 4.0 | 2.3 | 1.2 | 1.5 |
| End beam | 150 | 200 | 2.7 | NA | 1.8 | NA |
| Side sill (1) | 150 | 300 | 1.3 | NA | 1.2 | NA |
| Side sill (2) | 20 | 200 | 1.3 | NA | 1.2 | NA |

*Per component, including any other components that were modified to support load.
**Per component.

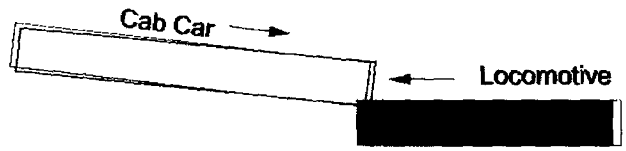
## EFFECTIVENESS OF SELECTED STRUCTURAL MODIFICATIONS

A dynamic model of an oblique collision between a cab car and a locomotive was used to evaluate the effectiveness of the structural modifications. Each vehicle has three degrees of freedom: translation in the longitudinal and lateral directions, and rotation about the vertical axis (yaw).

The model uses the force/crush behavior of the cab car's end beam and side sill to calculate the motions of the equipment. The locomotive was assumed to be rigid. The principal cab car structural members crushed during the collision were the end beam and the side sill. Because override did not occur, the vertical members - corner post, collision post, door post - were not loaded directly in the collision. There was little damage to the main structural elements of the locomotive; there was sheet-metal damage to the short hood and the cab and some bending of the plow. The model includes the track forces acting on the equipment. (Appendix B-2 contains a more detailed description of this model.)

The geometries of the two impacting structures, as well as the oblique angle between the cars, influence the direction of the forces between the cars. The magnitude of the lateral and longitudinal force components control the trajectories of the colliding cars during a collision. The motions of the cab car and locomotive are used to predict the crush of the cab car.

Figure 5a illustrates the predicted relative positions of the cars when the end beam fails. The large inertia of the locomotive makes it more or less follow the track (although for most cases analyzed, derailment of the locomotive is predicted), while the cab car yaws and begins to deflect away from the locomotive. Figure 5b illustrates the relative position of the cars when the lead corner of the locomotive has just lost contact with the side of the cab car, i.e., when the primary collision has ended. At this point, the cab car has yawed toward the locomotive, increasing engagement between the cab car and the locomotive. The end beam caused the initial yaw of the cab car away from the locomotive, tending to deflect the cab car and the locomotive past each other, while the side sill subsequently caused the yaw of the cab car toward the locomotive, increasing the engagement of the cab car and locomotive.




Note: dashed outlines represent initial positions.
Dimensions and displacements are approximate.

**Figure 5a: Cab Car and Locomotive Positions, at Initial Impact and at End Beam Failure.**

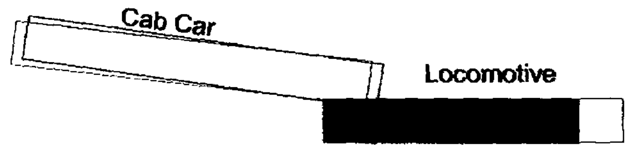



**Figure 5b: Cab Car and Locomotive Positions, at Initial Impact and at End of Contact.**

Table 5 lists the end beam and side sill ultimate strengths for the cases analyzed. For Case 1, the baseline case, the force required to crush the end beam is approximately 150,000 lbs and the force required to crush the side sill is approximately 400,000 lbs. (The force/crush characteristics of the end beam and side sill are shown in Appendix B.) The shape of the force/crush characteristics is assumed to remain the same as the baseline case, with the magnitude of the force changed appropriately. Case 6 corresponds to an end structure which incorporates the modified end beam and modified side sill (2) in Table 1. Two sets of cases have been analyzed: one with both the end beam and side sill strengthened and the other with only the end beam strengthened.

**Table 5: Cab Car End Structure Strength Cases Analyzed**

| Case | End Beam Ultimate Strength (kips) | Side Sill Ultimate Strength (kips) |
|---|---|---|
| 1 | 150 | 400 |
| 2 | 300 | 800 |
| 2A | 300 | 400 |
| 2B | 400 | 500 |
| 3 | 450 | 800 |
| 3A | 450 | 400 |
| 4 | 600 | 800 |
| 4A | 600 | 400 |
| 5 | 750 | 800 |
| 5A | 750 | 400 |

These strengths of the end structures analyzed encompass a wider range than the component modifications which have been developed.

Further design work would be required to develop component modifications for those cases in which the component strengths exceed those of the modified components described in Table 1. Such component modifications may require other changes to the cab car design, in addition to changes in the component cross-sections.

Figure 6 shows a plot of crush distance as a function of end structure strength, with the cab car initially traveling 18 mph and the locomotive initially traveling 53 mph in the opposite direction. The crush distance is the reduction in length along the impacted side of the cab car.

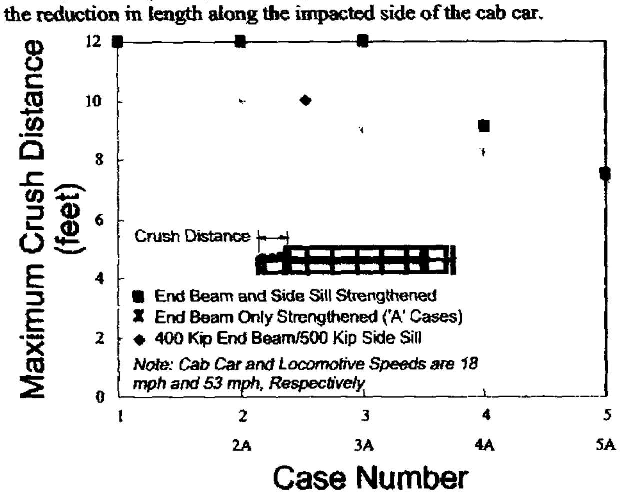



**Figure 6: Influence of Cab Car Strength on Cab Car Crush.**

The results in Figure 6 for the cases in which only the end beam is strengthened indicate that increasing the force required to crush the end beam increases the deflection of the vehicles out of each other's way. This deflection acts to reduce the crush of the cab car. In contrast, the increased side sill strength tends to increase the engagement of the vehicles, thus maintaining the 12 feet of cab car crush, until the end beam is strengthened by a factor greater than 3.

Figure 7 plots the maximum safe closing speed for operators and passengers, assuming that crush greater than 1 foot would reduce the operator's cab volume beyond the survivable limit, and that crush greater than 3 feet would begin reducing the passenger volume beyond the survivable limit for passengers seated near the operator's cab. The maximum closing speed in which the operator may be expected to survive is approximately 17 mph for Cases 5 and 5A and 9 mph for the baseline design (Case 1). The maximum closing speed in which all passengers may be expected to survive is approximately 32 mph for Cases 5 and 5A, with the end beam strengthened by a factor of five greater than the baseline, and 19 mph for the baseline case. Side sill strength does not influence the results depicted in Figure 7, as the side sill is not loaded for these closing speeds.

The relatively modest decrease in cab car crush (factor of two or less from baseline) for relatively large increases in end structure strength (factors up to 5 from baseline) is principally due to the kinematics of the collision. To a large degree the motion of the locomotive is unaffected by increases in cab car end strength. The locomotive lateral and yaw motions do increase favorably with increased cab car end structure strength, but this increase is modest. Motions of the cab car are influenced by its end-structure strength, however, these influences are not sufficient to allow the crush to decrease substantially.

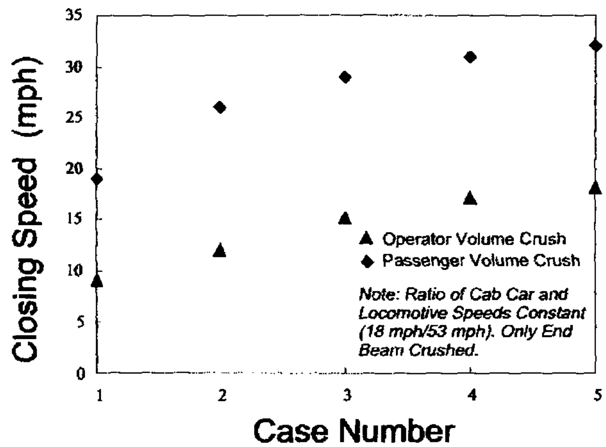



**Figure 7: Influence of Closing Speed on Maximum Cab Car Crush.**

Figure 8 shows a plot of crush distance as a function of end structure strength for three different closing speeds: 20, 35, and 50 mph. The plot presents the maximum crush for each of the five end structure design cases analyzed. As noted in the figure, the ratio of the cab car and locomotive speeds is the same as in the baseline case (18mph/53 mph). For an increase in closing speed by a factor of 2.5 (from 20 to 50 mph), the crush increases by a factor of four, from approximately 3 feet to 12 feet.

The influence of closing speed on cab car crush during the collision is principally due to the dynamics of the collision. In nearly all cases, there is significant crushing of the end beam, which means that the force exerted by the end beam is the same. The duration of time this force is acting between the locomotive and the cab car is greater for lower closing speeds. At lower closing speeds, the cab car and locomotive have more time to move out of each others way, resulting in less crushing of the cab car.

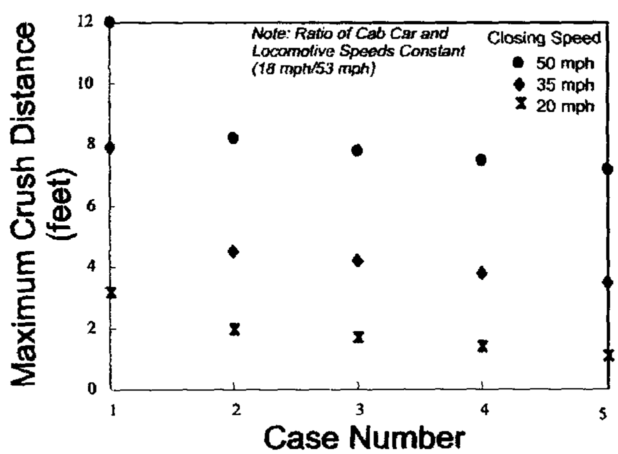



**Figure 8: Maximum Safe Closing Speed for Operators and Passengers.**

In both the 50 and 71 mph collisions involving the baseline design, the crush continues until the body bolster is loaded, which occurs after approximately 12 feet have been crushed. Owing to the body bolster producing a large increase in force for a small increment of crush, the cab car crushes up to the body bolster for a range of collision speeds.

## ANALYSIS CONCLUSIONS

For the oblique cab car to locomotive collision analyzed, a substantial increase in the strength of the cab car end structure results in a modest decrease in crush of the occupant volume. A modest decrease in the collision closing speed results in a substantial decrease in cab car crush.

In order to significantly decrease the severity of oblique collision consequences, conditions in which cab car occupants are expected to survive, structural designs significantly different from existing designs will be required. These structures may benefit from having significantly different geometries from current designs, and may benefit from allowing more structural crush without occupant volume intrusion than current designs.

The increase in crashworthiness performance is modest in comparison with the increase in cab car end structure strength, however, the increase in weight and cost is also modest for such an increase in cab car end structure strength. Evaluation of the weight of potential modifications to existing cab car structures indicates that modifications of cab car end structures to increase end beam, corner post, door post, and side sill strength would add ~670 lbs (~0.7%) to the total weight of the car. Such modifications would add approximately $2000. (~0.1% for purchase price of $1,500,000.) to the purchase price of a cab car.

## ACKNOWLEDGEMENTS

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## APPENDIX A: ACCIDENT REVIEW

On February 9, 1996, the cab car of a New Jersey Transit (NJT) commuter train made up of a cab car, four coach cars, and a locomotive, struck the locomotive of another NJT commuter train, made up of a locomotive, five coach cars and a cab car. There were three fatalities: the locomotive operator, the operator of the leading cab car, and a passenger in the same cab car. There were twelve serious injuries, all in the leading cab car.

The main structural elements (i.e., body bolster, center sill, draft sill) of the cab car in the Secaucus collision remained essentially intact. The structural damage consisted principally of crushing of one side of the vehicle, from the end beam to the body bolster longitudinally, and from side sill to the roof sill vertically. There was significant damage to the impacted corner post, end beam and side sill. The end beam failed near the base of the collision post. Figure 9 schematically illustrates the structural damage to the frame of the cab car.

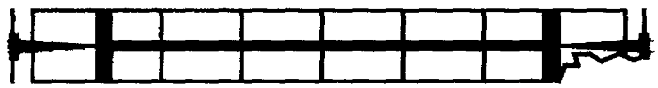

**Figure 9: Secaucus Underframe Collision Damage.**

Seat frames in the cab car were crushed or missing from the body bolster forward on the side of impact. In areas away from the structurally damaged sections, the seat frames and luggage racks generally remained intact.

Figure 10 illustrates the sequence of events during the collision, based on conversations with FRA and NTSB officials. It appears that the collision progressed as follows:

1.) The cab car was traveling at approximately 18 mph when it struck the front, right corner of the locomotive, which was traveling at approximately 53 mph in the opposite direction. Based on the track geometry, the angle between the two vehicles at the instant of impact was approximately 7°. The corner post on the right side of the cab car struck the right side of the locomotive. Both collision posts on the cab car remained in place, though the right post incurred some structural damage. The right corner post was torn away from the cab car. The roof plate from the right side of the cab car broke away and penetrated the window of the locomotive.

2.) The cab car raked down the side of the locomotive. The left rail (field side) under the locomotive rolled over and the locomotive derailed.

3.) The derailed locomotive pulled the trailing cars off the track. The cab car continued to rake the cars trailing the locomotive, damaging stair wells and radius rods as it went.

4.) Most cars in the locomotive-led consist derailed (the last car may have stayed on the track). Only the cab car derailed in the cab car-led consist. The cab car-led consist was stopped by the collision at the switch. The locomotive-led consist slid to a stop on the ties and ballast.

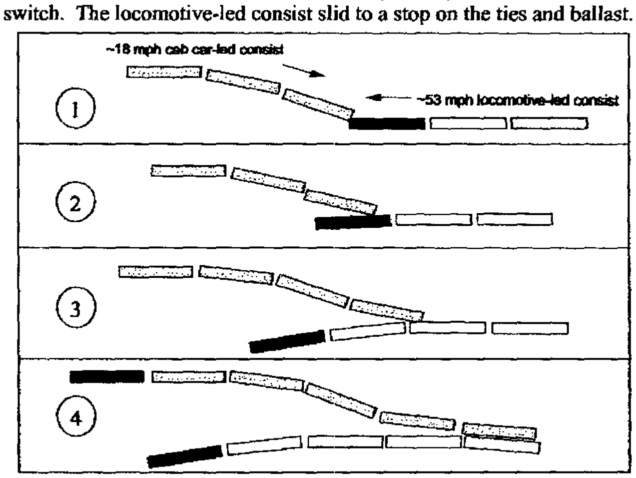


**Figure 10: Schematic of February 9, 1996, Secaucus, New Jersey, Cab Car/Locomotive Collision**

Damage occurred principally to the lead vehicles of each of the trains. Damage also occurred to the stair wells and the radius rods of the remaining cars in the locomotive-led train due to the scraping of the two trains past each other. The lead locomotive was a GM-built GP40PH-2, rebuilt by Morrison-Knudsen in 1993. The lead cab car was built by Bombardier with a steel underframe and aluminum superstructure.

The roof plate of the cab car penetrated the operator's window of the locomotive. Damage to the hood of the locomotive appeared to be due to the roof plate riding up on the hood and through the window. Some superstructure damage to the front of the locomotive, approximately halfway between the coupler and the side of the locomotive, appeared to have been caused by the front of the cab car. The center sill and main structure of the locomotive remained essentially intact. A post-collision photograph of the locomotive is shown in Figure 11.

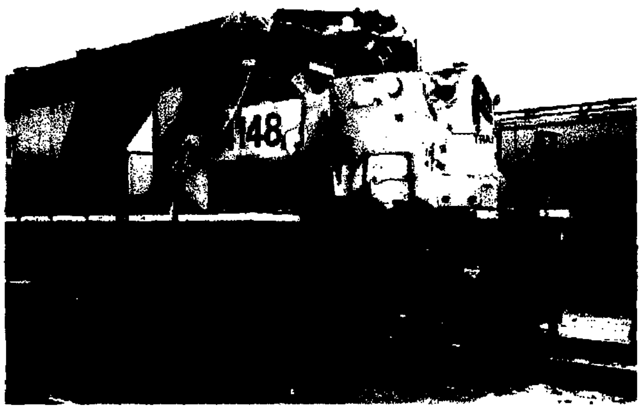


**Figure 11: Lead Locomotive, NJT Train.**

Damage to the lead cab car includes crushing of the right, front corner of the car from the end of the car to the body bolster. This area includes the operator's compartment and approximately five rows of seats. The right (track-side) collision post incurred substantial damage: there are several large cracks in and around the attachment point. This damage may have occurred when the end of the transverse floor member was torn off in the initial collision with the locomotive. The collision post itself may not have been loaded directly. The corner post and a portion of the roof plate were separated from the cab car. A post-collision photograph of the cab car is shown in Figure 12.

**Figure 12: Lead Cab Car, NJT Commuter Train.**

Substantial damage was also incurred by the seats across the aisle from the crushed seats, due to debris from the collision and damage to the floor. In all, approximately 25 seat positions were destroyed during the collision. The remaining seats -- those not directly crushed by the collision – appeared to be essentially intact. Approximately two seat-pairs were out of position, however, they may have been put out of position by rescue or cleanup personnel.

## APPENDIX B: COLLISION DYNAMICS MODELS

This Appendix presents descriptions of a simplified rigid body model which describes the fundamental mechanics of the car motions during an oblique collision and a more detailed lumped mass model appropriate for determining the influence of cab car end structure modifications on the consequences of an oblique collision between a cab car and a locomotive, such as occurred at Secaucus, NJ in February 1996. Both models are implemented in Mathcad® worksheets.

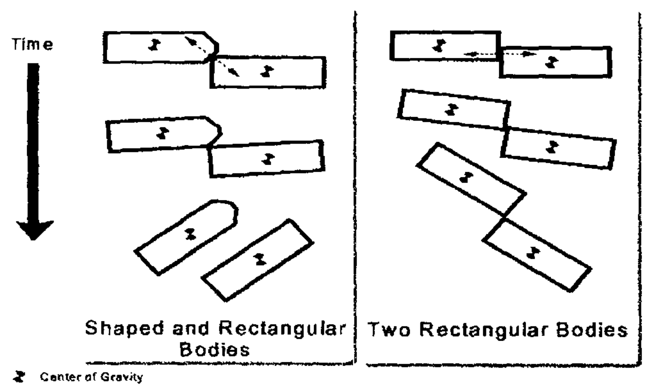


**Figure 13: Time-History Schematic of Oblique Collision of Two Rigid Bodies.**

The undeformed and deformed geometries of the carbody structures influence the direction of the forces, including the magnitude of the lateral force component, consequently influencing the gross motions of the colliding cars during a collision. Figure 13 illustrates the influence of geometry on the results of an offset collision between two rigid bodies. For rectangular rigid bodies, the impact force causes both bodies to turn into each other (both bodies turn clockwise in the illustration). In a collision between a shaped body and a rectangular body, the bodies turn away from each other.

### B.1 RIGID BODY COLLISION DYNAMICS

Figure 14 is a schematic of a shaped body (cab car) colliding with a rectangular body. The general plane motions of these bodies can be determined from rigid body mechanics, assuming a perfectly plastic collision and conservation of momentum. Table 6 lists the parameters of the rigid body model.

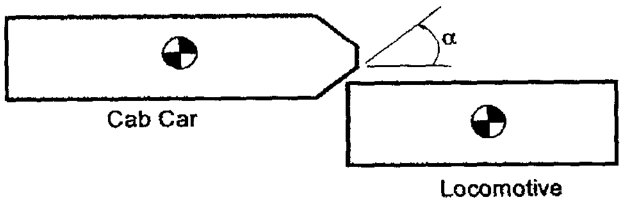


**Figure 14: Model of Oblique Collision of Two Rigid Bodies.**

**Table 6: Parameters of Rigid Body Model.**

| Parameter | Cab Car | Locomotive |
|---|---|---|
| Weight | 100 kips | 260 kips |
| Yaw Inertia | $1.896 \times 10^6$ ft-lb-second$^2$ | $2.49 \times 10^6$ ft-lb-second$^2$ |
| Length | 85 feet | 60 feet |
| Width | 10 feet | 10 feet |

Figure 15 shows the influence of the cab car end structure geometry angle α for a collision in which the locomotive is initially standing and the cab car is moving with a longitudinal velocity of 30 mph. The centers of gravity of the two cars are offset laterally by 10 feet. For cab car end structure angles less than approximately 8 degrees, the cars turn into each other, i.e., both cars turn clockwise, while for cab car end structure angles greater than 8 degrees, the cars turn away from each other, i.e., counterclockwise.

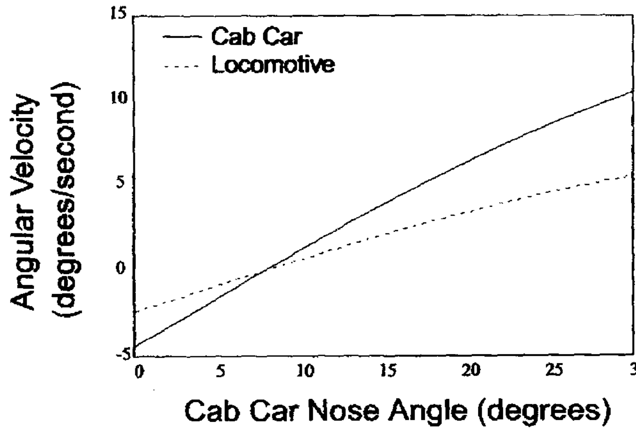


**Figure 15: Influence of Cab Car Nose Angle on Cab Car and Locomotive Yaw Velocity, Rigid Body Model.**

### B.2 DEFORMABLE BODY COLLISION DYNAMICS

Figure 16 is a schematic of the oblique collision dynamics model. The model allows for the general plane motion of the cab car and the locomotive. The forces acting on the cars include the track forces and the forces due to crushing of the cab car end structure. The locomotive is assumed to be rigid relative to the cab car. Coupler forces are neglected. Due to the location and nature of the draft gear, the coupler cannot develop a significant yaw moment about the center of gravity of the car, and consequently does not significantly influence the gross motions of the impacting cars. In addition, there was minimal, if any, damage to the trailing draft gears of the locomotive or cab car and their associated coupled cars in the Secaucus collision. The lack of damage indicates that the coupler force levels were relatively low.

The force/crush characteristic used in the collision dynamics model was developed from end beam and side sill force/crush characteristics calculated with a finite element model, and the body bolster behavior inferred from previous analysis results. The end beam, side sill, and body bolster are loaded sequentially in the collision dynamics analysis.

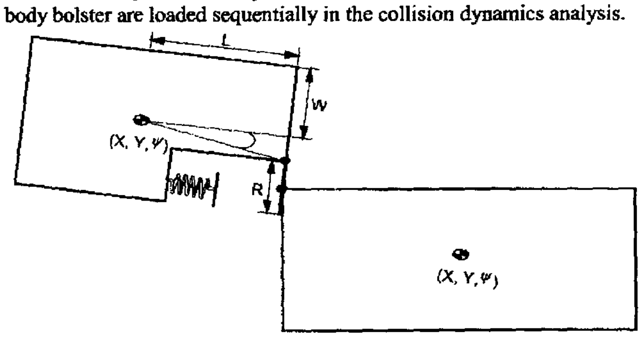


**Figure 16: Schematic of Oblique Collision Dynamics Model.**

Force/crush characteristics for the end beam and the side sill were developed using a finite element mesh of the entire vehicle length, with the longitudinal midplane treated as a plane of symmetry (Mayville, et al, 1996). This means that the model of a collision or corner post load was a simulation of loading both collision or corner posts. The mesh was constrained from displacement only at the bolster locations. The load was applied quasi-statically to the components through a simulated rigid, cylindrical body as illustrated in Figure 17. In the figure, the load shown is being applied to the corner post 18 inches above floor level, ahead of the end side door.

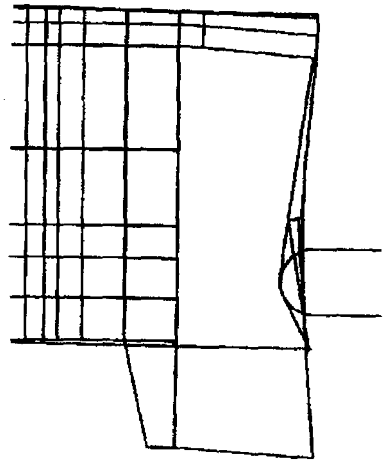

**Figure 17: FEA Mesh with Load Applicator.**

The force/crush characteristics for the side sill aft of the end door for the baseline and modified designs is shown in Figure 18. The side sill is loaded longitudinally, initially just aft of the end side door at floor level. For the baseline side sill, the yield strength calculated is approximately 240 kips, and the ultimate strength calculated is approximately 400 kips. For the modified designs, the yield strength is 300 kips and the ultimate strength is 500 kips. The nature of the force/crush characteristic is essentially the same for the baseline and modified side sill designs.

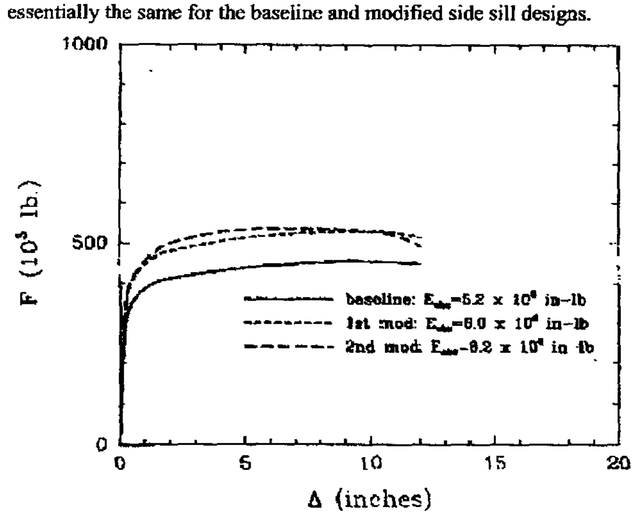


**Figure 18: Modified and Baseline Side Sill Force-Crush Characteristics at Base of Door Post.**

Figure 19 shows the force/crush characteristic for the end beam when loaded at its outboard edge, near the sidewall just ahead of the end side door. For the baseline end beam design, the yield strength is approximately 110 kips, and the ultimate strength is approximately 150 kips. For the modified design, the yield strength is 300 kips, and the ultimate strength is 400 kips.

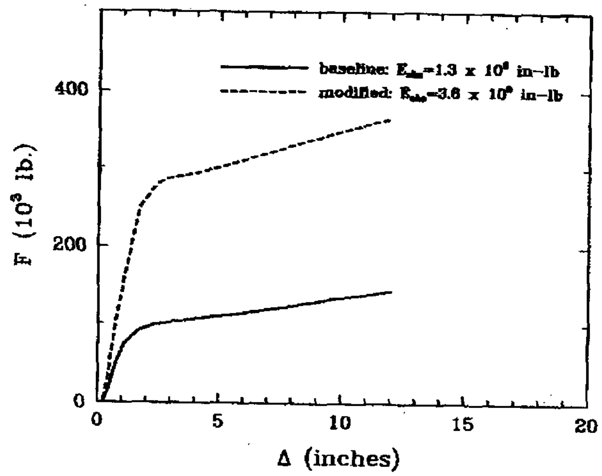


**Figure 19: Modified and Baseline End Beam Force-Crush Characteristics at Base of Corner Post.**

Results from previous analysis indicate that the body bolster may sustain longitudinal loads near the sidewall of the car up to approximately 800 kips before there is significant structural damage aft of the body bolster. (There was little, if any, damage to the structure aft of the lead body bolster of the cab car involved in the Secaucus collision.) Previous analysis results indicate that a longitudinal load of at least 1600 kips applied to the draft stops is required to cause more than 6 inches of crush of the main structure (Mayville, et al, 1996).

Current analysis results show that a load of approximately 400 kips is required to significantly crush the side sill (see Figure 18) ahead of the body bolster. The side sill cross-section is the same ahead and behind the body bolster, and consequently it is likely that it will crush at the same load aft of the body bolster as it does ahead of the body bolster. Aft of the body bolster, this load is principally carried by the center sill and two side sills, as shown in the free body diagram for the load applied at the draft stops in Figure 20. When a load of 800 kips is applied to the body bolster at the side wall of the car, the side sills and center sill are carrying loads near those required to cause significant crush.

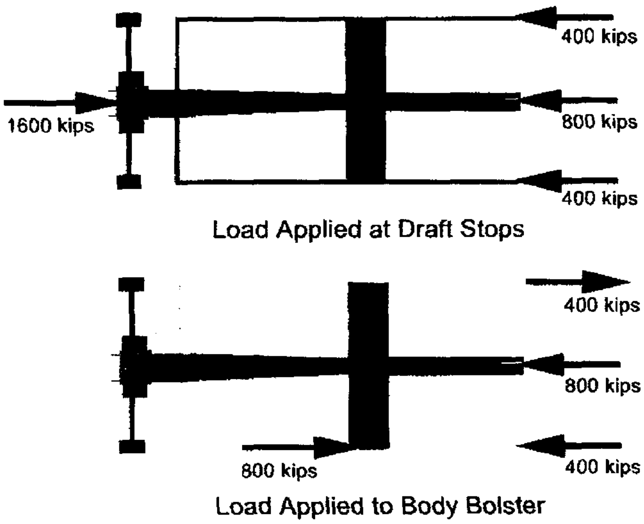


**Figure 20: Car Structure Free-Body Diagrams, Load Applied at Draft Stops and Load Applied at Body Bolster Near Car Sidewall.**

In the train collision mechanics model, the end beam, side sill, and body bolster are loaded sequentially. The end beam is assumed to behave like a rigid beam with the force/crush characteristic shown in Figure 19. A greater load, resulting in the same moment about the connection of the end beam to the draft sill, is required when the load is applied inboard from the sidewall of the car. The load transverse to the end beam is assumed to be small compared with the load normal to the beam. The load applied to the side sill and body bolster is assumed to act in the longitudinal direction of the car, i.e., the transverse loads supported by these members are assumed to be small compared with the longitudinal loads. Loads applied to the body bolster are assumed to increase linearly to 800 kips in 6 inches from the location of collapse of the side sill. The force crush characteristic for the baseline vehicle for a load applied near the sidewall of the car is shown in Figure 21.

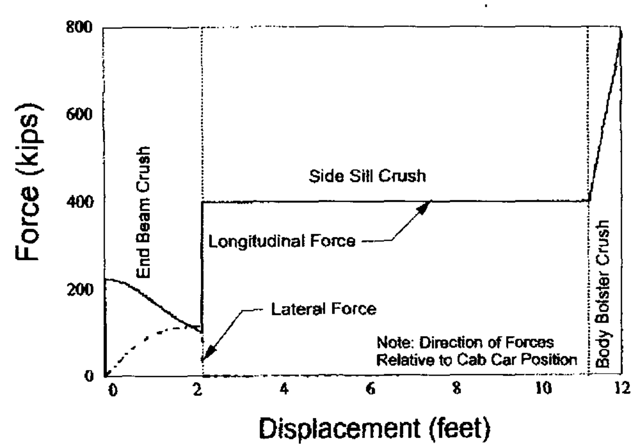


**Figure 21: Baseline Cab Car End Structure Force**

**Displacement Characteristic.**

Figure 22 shows the lateral force acting on the carbody at body bolster as a function of body bolster displacement from track centerline. Such a force acts on both body bolsters. This force is assumed to be proportional to displacement, until the displacement has become sufficient to indicate derailment. Derailment is taken to occur at an L/V ratio of 0.6. Subsequent to derailment, the flanges of the wheels are presumed to plow the ballast. The L/V ratio is assumed to be 0.6 for the wheels laterally plowing the ballast.

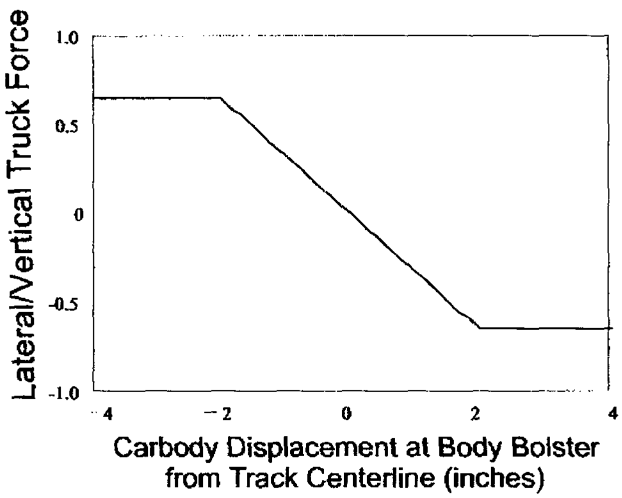


**Figure 22: Lateral Force Acting on Carbody at Body Bolster.**

The track layout approximates an AREA No. 15 Turnout (Manual for Railway Engineering, 1972). This layout geometry is illustrated in Figure 23. The locomotive is positioned on the tangent track, while the cab car is positioned on the curved track.

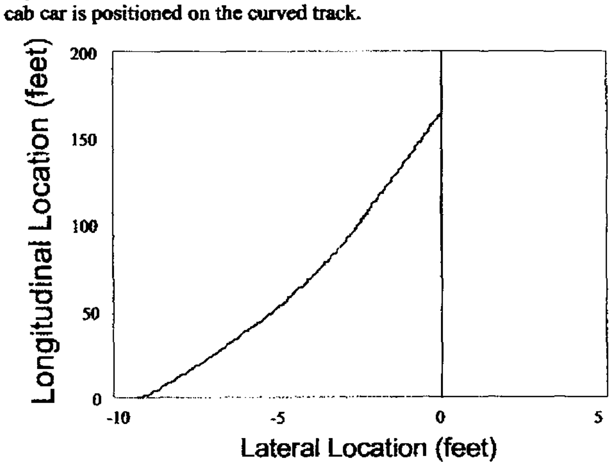


**Figure 23: Track Layout Geometry.**